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STRESS ANALYSIS OF A 100-INCH (CW-351) PLASTIC POLYPROPYLENE DO--ETC(U)
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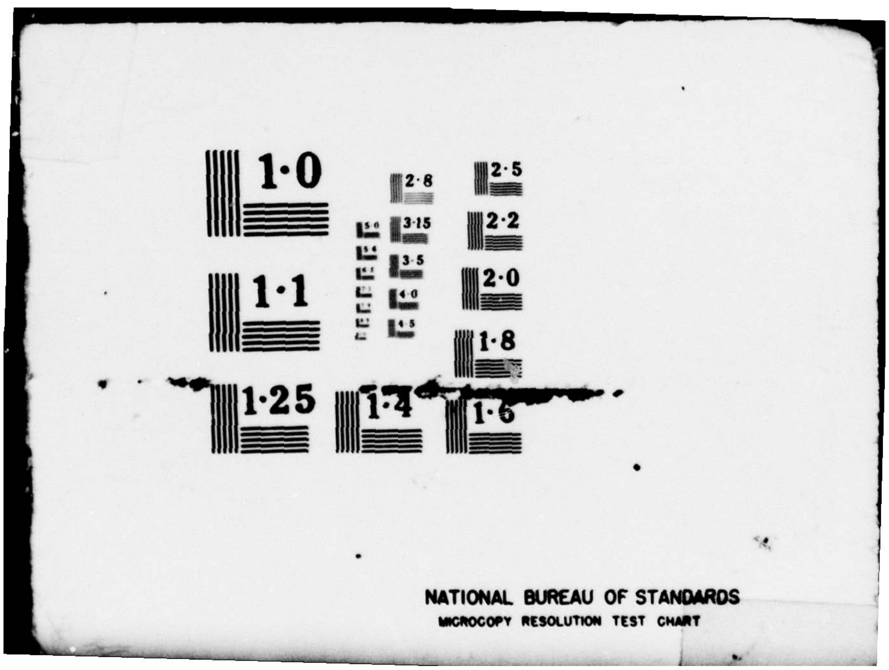
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U. S. NAVY UNDERWATER SOUND LABORATORY
FORT TRUMBULL, NEW LONDON, CONNECTICUT

STRESS ANALYSIS OF A 100-INCH (CW-351) PLASTIC POLYPROPYLENE DOME

By

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Technical memo.

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BACKGROUND

The use of plastic in the fabrication of sonar domes is not new. For submarine applications, plastic domes are used to cover the JT hydrophone, reference (a). For surface vessels, a considerable study has been performed both on the material and the fabrication aspects of sonar domes, references (b), (c), and (d). For the AN/SQS-17 system, a fiberglass dome was fabricated, and it performed adequately under service conditions, reference (e).

The Laboratory, as part of its general program interest in sonar domes, reference (f), is investigating the use of a polypropylene dome. The inexpensiveness of the polypropylene raw material and fabrication, together with polypropylene having a specific acoustic impedance close to that of water, 1.9×10^5 versus 1.5×10^5 gm-cm/sec^2 for water has given impetus to such a study.

INTRODUCTION

Sonar domes fabricated from steel are the standard naval type. These domes consist of a thin shell covering supported by a punched-out backing up plate. A rod truss-work provides the necessary stiffness for the complete dome structure.

In utilizing a plastic dome, the main source of strength must lie in the skin, with the possible addition of rectangularly shaped stiffeners. The AN/SQS-17 dome, reference (e) made of fiberglass (compressive strength $\cong$50000 psi; tensile strength $\cong$50000 psi and elastic modulus $\cong 3 \times 10^6$ psi), has no stiffeners and the skin thickness is 7/16 inch.

The opinions expressed are those of the author(s) and not necessarily the official views of the Laboratory.

Acoustic considerations dictate that the skin thickness of the polypropylene dome be 9/16 inch thick; however, a standard sheet thickness of 5/8 inch would be used in the actual fabrication.

Physical Properties of Polypropylene

The following properties of polypropylene were obtained from the literature, reference (g), or from tests performed for the Laboratory, on tensile and compressive specimens, reference (h).

$$
\begin{aligned}
&\text{Tensile strength} \approx 5000 \text{ psi *}\\
&\text{Compressive yield strength} \approx 5000 \text{ psi *}\\
&\text{Elastic modulus} \approx 2 \times 10^5 \text{ psi}\\
&\text{Poisson's ratio} = 0.45\\
&\text{Specific gravity} = 0.91\\
&\text{Impact strength (Izod)} = 1 \text{ ft-lb/inch of notch}\\
&\text{Hardness} = \text{75D, shore durometer, or R-95}
\end{aligned}
\qquad (1)
$$

Tensile and Compressive tests of polypropylene have shown the material to be elastic up to the yield point.

Types of Loadings on the Dome

Before analyzing the structural characteristics of the dome, the types of loading expected must be determined. The sonar dome is attached unto the keel of a DE or DD class of surface craft. Because of the maneuvers and seakeeping qualities of such ships, the following types of loadings can be encountered:

Type A - Normal drag forces during a zero degree yaw, 35 knot speed run;

Type B - Sideways form drag forces during a turning maneuver at 35 knots in which the yaw angle will be ten degrees;

Type C - Lateral loading on dome side caused by ship rolling;

Type D - Impact forces or "slam" occurring from the quenching of the dome, and

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* An allowable working stress of 3600 psi will be used in the analyses. This is a factor of safety of 1.39.

Type E - Inertial loading of the dome from the mass of entrapped water.

These loadings can occur in pairs simultaneously, for example, Types A and C, or B and C. If the transducer is supported by the dome (not recommended, however), then both static and inertia loads originating with the transducer must be considered.

Structural Components of the Dome

There are four main structural components of the dome. Each component will be acted upon by one or more of the previous types of loadings.

Referring to Fig. 1, the structural areas together with the types of loadings are as follows:

Area I - Front of Dome This area will have to resist Types A or D or E.

Area II - Bottom of Dome This area will have to resist Type D.

Area III - Sides or Body of Dome This area will have to resist Types B and C, or D, or E.

Area IV - Mounting flange for Bolting This area will have to resist Types B and C, or E.

Previous Analyses or Studies on Dome Strength

Reference (b) has a short analysis of a laterally loaded plate with simply supported edges. Reference (c) contains experimental data on the edgewise compression stresses and outlines an analysis previously performed in which the dome is treated as an equivalent cylindrical surface. A uniform load of 25 psi was considered. Reference (i) is a valuable study on the stress characteristics of steel domes. In addition, pressure diagrams obtained from reference (j) and discussed qualitatively below, are shown along the dome contour.

The information from references(i) and (j) has been used in the present analysis.

Pressure Distributions on the Dome

Reference (j) reports on wind tunnel tests performed on four domes of 100 inch length but different profiles through the transverse sections. For the fleet type shape, the so-called EPH type sonar dome

(CW-351), the following qualitative results are shown:

A. <u>At the Keel Line of the Dome</u>

a. Pitching for a constant yaw angle, has very little effect on the pressure distribution;

b. At the "forefoot", i.e., the intersection of the keel line with the front of the dome, at any yaw angle, the pressure varies from a small positive value to a small negative value as the pitching angle changes from upward 5 degrees to downward 5 degrees. A short distance aft of the front point, a large negative pressure results; this point being the more critical in terms of incipient cavitation;

c. The remainder of the keel line exhibits negative pressure. This pressure is equivalent to an internal pressure loading on a shell structure.

B. <u>At the Joint of the Body and Bottom of the Dome</u>

a. Pitching, for a constant yaw angle, has very little effect on the pressure distribution;

b. At all yaw and pitch angles, negative pressures exist through the joint line, except for zero or small positive pressures at the front for upward pitch angle configurations.

C. <u>Transverse Middle Line of Body</u>

a. At zero yaw, change in pitch from upward 5 degrees to downward 5 degrees has little effect on the pressure distribution.

b. At 5 and 10 degrees of yaw at a constant pitch, marked differences in pressure were obtained on the port and starboard sides of the dome. The larger negative pressure was on the leeward side of the flow.

c. At all angles of yaw, the front of the dome has positive pressure loading. The stagnation point lays approximately at the point between 0 and 10 percent chord. The rest of the body had negative pressure loading.

ANALYSIS

The dome structure is not a true body of revolution. Additional "hard spots" or stiffener points are introduced at baffle locations.

Discontinuity stresses because of changes in curvature also occur at the junction of the bottom with the side of the dome. These stresses are evaluated in the analysis. To attempt an exact analysis of the dome, much valuable effort will be lost. The author has set up the differential equations defining the equilibrium condition of a semi-ellipsoidal shell, but the solution to such an equation can only be attempted numerically with the aid of a computer. And, in the end, the results will cover only a small aspect of the problem. Consequently, the following analysis is of an engineering nature, very conservative in the values of loading and allowable stresses used. If the structure is capable of resisting the "paper" values set up in this memorandum, then it should be strong enough to perform adequately in service. *It is assumed that the loads are of short duration, thus the effect of creep is not taken into account in the analysis. The question of creep, however, is serious; and actual use of the dome is necessary to resolve any doubts concerning the usefulness of the polypropylene material.

Area I - Front of Dome

The front portion of the dome can be represented as a circular cylindrical shell with an equivalent radius of 9 1/2 inches and a length of 50 inches, Fig. 1. From reference (i) it is found that a maximum dynamic pressure of 32 psi (negative) exists at the bottom of the dome at 0-degree yaw for a speed of 35 knots. A dynamic pressure of p = 40 psi (negative) will be used for all calculations where applicable.

Membrane Stresses in Front of Dome

Assuming that the bending stresses are negligible, the tangential shell stress, σ_T, is : (reference (k))

$$\sigma_T = \frac{pR}{t} \qquad (2)$$

where p = external pressure (negative), equivalent to an internal pressure;

R = radius of cylinder;

t = thickness of shell

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* This statement obviously doesn't hold if the vessel were to become aground on the dome.

Inserting values into eq. (1);

$$\sigma_T = \frac{40\ psi\ (9.5\ in)}{5/8} = 676\ psi\ (Tension) \tag{3}$$

From eq. (1), the tensile ultimate is seen to be 5000 psi. Therefore, the front of the dome will have a factor of safety of approximately 9 with respect to the tensile strength. The allowable working stress is 3600 psi, and for this stress, the factor of safety is 5.3.

Critical Skin Buckling Load - Front of Dome*

At a speed of 35 knots the maximum positive pressure on the front of the dome, reference (*l*) is 18 psi. A pressure loading of 20 psi will be used for the design analysis.

The critical buckling load, P critical, of a uniformly loaded unstiffened circular cross-sectional arch; simply supported at its ends, is (reference (k)):

$$P_{critical} = \left(\frac{4\pi^2}{\phi^2} - 1\right) \frac{D}{R^3} \tag{4}$$

where

ϕ = central angle in radians = π, (see Figure 1);

D = stiffness of arch = $Et^3/12(1-\nu^2)$

t = thickness of shell, 5/8 inch;

E = elastic modulus, 2×10^5 psi;

ν = Poisson's ratio, 0.45

Inserting the values into equation (4):

$$P_{critical} = 18\ psi \tag{5}$$

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* The stiffening effect of the internal fluid is neglected.

The critical buckling stress of 18 psi is short of the actually assumed applied load of 20 psi*. The buckling stress difference in this calculation is marginal, differing only by 2 psi between strength capability and the assumed applied load of 20 psi. However, eq. (5), the critical buckling stress is exactly equal to the actual applied load at 35 knots, of 18 psi, reference (i). The equivalence of loads would indicate no danger to the front of the dome. If the maximum speed considered possible were reduced from 35 knots to 30 knots, the applied pressure (varying directly as the square of the speed) is reduced to 15 psi. This value is certainly below the critical stress of 18 psi.

Area II - Bottom of Dome

The bottom of the dome will have to resist the type D or E loading condition. The inertia loading, Type E will be considered first.

Type E - Inertia Loading on Bottom of Dome

Inertia loading on the bottom of the dome would arise from the following assumed set of circumstances:

a. The ship is in a logging condition with the sonar dome exposed from the sea. The dome itself has not been drained of sea water;

b. the sonar dome section of the ship accelerates downward through the air; however, before the dome is submerged in the water,

c. the ship's momentum is stopped by the action of another wave front, consequently,

d. the entrapped sea water in the sonar dome "pushes" against the bottom of the dome.

The maximum distributed load on the bottom of the dome is obtained as follows:

The height of the domes is 50 inches = 4.16 feet. The uniform load is consequently 64.4 psf times 4.16 feet, or 268 psf or 1.86 psi. Assuming that the maximum acceleration is of 1g (this value is reasonable, reference (m)), then the total applied load will be 2 x 1.86 psi $\approx$ 4 psi.

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* To compare these values with the AN/SQS-47 dome, reference (e). For that dome, $t = 7/16$ inch; $E = 3 \times 10^6$ psi, $\phi = \pi$ radians, $\nu = 0.45$ and $R = 12$ inches, so that $f_{critical} = 56$ psi. The applied load is again assumed to be 20 psi.

Membrane Stresses in Bottom of Dome

The bottom of the dome is assumed to be a semi-ellipsoidal shell, major axis of 100 inches and minor axis of 40 inches.

The maximum membrane σ_θ stress in an ellipsoidal shell is given by (reference k):

$$\sigma_\theta = \frac{p a^2}{2bt} \qquad (6)$$

where p = applied internal pressure = 4 psi;

a = semi-major axis = 50 inches;

b = semi-minor axis = 20 inches;

t = thickness of the shell = 5/8 inch

solving for equation (6):

$$\sigma_\theta = 400 \text{ psi (Tension)} \qquad (7)$$

With an allowable working stress of 3600 psi, the factor of safety for this type of loading is: 11.

Slam Load on Bottom of Dome

Reference (n) outlines the relevant and more recent theories concerned with ship slamming. For the design of the bottom of the dome, an impact coefficient technique will be used. The impact coefficent, C_I, is defined as:

$$C_I = F / \tfrac{1}{2} \rho d V^2 \qquad (8)$$

where:

F = force on bottom of dome per unit length of dome (total force assumed to act uniformly);

ρ = density of water;
d = diameter of an equivalent cylinder;
V = velocity of impact as cylinder strikes the water.

From Fig. 6 of reference (n), impact coefficients derived by various theories and experimental results are shown. For a NACA TN 2889 section, the maximum C_I value experimentally obtained is approximately $1.2 \times \pi$ or 3.77. The equivalent cylindrical diameter of the dome is assumed 40 inches or 3.33 feet. The impact velocity of the dome is obtained as follows:

From reference (m), a severe pitching angular velocity, ω_o , is shown to be on the order of 1 radian/sec. Assuming a single degree of freedom system, and assuming that the centroid of the dome is 50 feet from the ship's center of pitching, then the tangential velocity of the dome upon impact with the water is 1 rad/sec times 50 feet or 50 feet/sec. This value of velocity ignores the viscous damping effect exerted upon the aft portions of the hull as the ship enters and leaves the water. Consequently, the tangential velocity of the dome would be considerably lower. If the ratio of the coefficient of damping to the critical damping is assumed to be 90 percent, then the assumed angular velocity of the dome would be:

$$\omega = \omega_o \sqrt{1-\left(c/c_c\right)^2} = \omega_o \sqrt{1-0.81} = 0.44\,\omega_o = 0.44 \text{ rad/sec} \tag{9}$$

Hence the velocity, V, of impact for the dome would be: 0.44 x 50 feet, or 22 feet/sec. The impact force from equation (8) becomes:

$$F = C_I \frac{\rho}{2} d V^2 = 3.77(1)(3.33\,ft)(22\,ft/sec)^2 = 6074 \text{ lbs/ft} \tag{10}$$

The length of the dome is 8.35 feet, thus the total impact force is: 8.35 x 6074, or 50414 pounds. The area of the dome bottom is approximately 40 inches times 100 inches, or 4000 square inches. The impact pressure becomes:

$$p_{impact} = \frac{50414 \text{ lbs}}{4000 \text{ in}^2} = 12.7 \text{ psi} \tag{11}$$

However, for design purposes an impact pressure of 1000 psf or 7 psi will be used.* The equation for the critical buckling load of the dome bottom is similar to eq. (4) viz:

$$p_{critical} = \left(\frac{4\pi^2}{\theta^2} - 1\right) D/R^3 \tag{12}$$

*In conversation with Mr. K. H. Wilcoxon, Code 553A, David Taylor Model Basin, it was indicated that this figure has been used for the design of domes.

where θ = central angle = 140° = 2.44 radians;

R = radius of dome section = 20 inches*

Solving equation (12):

$$p_{critical} = 3.6 \text{ psi} \tag{13}$$

The critical pressure load for the dome bottom is lower than the assumed applied load of 7 psi. The bottom of the dome is a non-critical area with respect to sonar activity. The placing of stiffeners there will in no way affect sonar operation. The bottom, in addition, will have an application of sound absorbing material. The sound absorbing material, by covering the stiffeners, will reduce any deleterious reflection effects arising from the stiffeners.

To protect the bottom of the dome from buckling, it is recommended that 1/2 inch by 1/2 inch stiffeners be placed every 12 inches, both laterally and longitudinally.

Discontinuity Stresses at Junction of Bottom and Side

Because of the change in curvature at the line in which the dome bottom meets the side, discontinuity stresses will occur. Reference (k3) treats the case of a shell having as its end an ellipsoid of revolution. The maximum stress at the joint (for a ratio of major axis to minor axis of ellipse equal to 2, approximately the value in the dome) is given as:

$$\sigma_{max} = 1.128 \; a \, p/t \tag{14}$$

where: a = semi-major axis = 50 inches

p = applied internal pressure = 10 psi**

t = thickness of shell = 5/8 inch

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* The dome bottom is doubly curved, see Fig. 1. The effect of the longitudinal curvature is added stiffness, since the lateral radius decreases. Equation (12) leads to a conservative value of critical stress.

** Reference (i) shows the pressure distribution at the bottom keel of the dome. An average uniform internal pressure is on the order of 10 psi.

Substituting the values into equation (14):

$$\sigma_{max} = 900 \text{ psi} \tag{15}$$

The discontinuity stress is not excessive, giving a factor of safety of 4. As a further insurance against failure at that joint, a stiffener from the dome bottom system will be placed at the junction of the bottom and side. The effect of the stiffener is to reduce the strains at the junction, hence the stress.

Area III - Sides of Domes

The sides of the dome must resist a normal pressure, or the inertia load transferred from the dome bottom, or the impact load also transferred from the dome bottom. All three conditions will be examined.

Normal pressure on Side of the Dome

From reference (i), it is seen that the sides of the dome experience a negative pressure from 0 to 10 degrees yaw. The maximum pressure is on the order of 23 psi at 35 knots. A design pressure of 25 psi will be used.

Membrane Stress in Dome Side Caused by Internal Pressure

Assuming the sides of the dome to be a portion of an ellipsoidal shell, the membrane stress is expressed as eq. (6)

$$\sigma = pa^2/2bt \tag{16}$$

where p = equivalent internal pressure = 25 psi,

a = semi-major axis = 50 inches

b = semi-minor axis = 20 inches

t = thickness of shell = 5/8 inch

Substituting into eq. (14),

$$\sigma = 2500 \text{ psi (Tension)} \tag{17}$$

The stress of 2500 psi (tension), eq. (17) is below the working stress of 3600 psi. The factor of safety associated with this loading is 1.4.

Lateral Loading on Dome Side Caused by Ship Rolling

Type C force previously discussed, is a sideways form drag force caused by the rolling motion of the ship. This force can be evaluated by the expression:

$$F_D = C_D \frac{\rho}{2} A V^2 \tag{18}$$

where:

F_D = drag force assumed to act uniformly;

C_D = coefficient of drag = 2

A = side area of dome = 50 inches x 100 inches = 5000 in.

V = maximum velocity at dome centroid

The centroid velocity is obtained as follows:

From reference (m), a severe value of rolling frequency is on the order of 1 rad/sec. Assuming that the centroid of the dome is 10 feet from the center of rotation, the velocity (assuming harmonic oscillation) is:

$$V = 10\,ft \cdot 1\ rad/sec = 10\ ft/sec \tag{19}$$

Substituting into eq. (13), there is obtained:

$$F_D = 6400\ lbs \tag{20}$$

Assuming the drag force uniformly distributed over the side of the dome, the pressure loading becomes from eq. (20):

$$p = 6400\ lbs / 5000\ in^2 = 1.28\ psi \tag{21}$$

The critical membrane stress from the loading of equation (21) is far below the value previously obtained in equation (17) from a loading of 25 psi.

However, the 1.28 psi being externally applied, may cause buckling. Assuming the dome side to be an equivalent circular arch of 90 inches diameter, see Fig. 1, wall thickness of 5/8 inches, as from reference (k). The critical buckling pressure is equation (12) where $\theta = 52° = 0.91$ radians:

$$p_{critical} = \left(\frac{4\pi^2}{\theta^2} - 1\right) D/R^3 \tag{22}$$

or

$$p_{critical} = 0.34 \text{ psi} \tag{23}$$

The analysis indicates that the dome is understrength. This loading could become particularly critical if the pressure due to roll were simultaneously applied with a pressure load from a yaw or turning condition.

Tensile Stress in Dome Side Caused by Inertia Loading

The inertia load was on the order of 4 psi, eq. (6). The total load is obtained by multiplying the uniform load by the total area of the bottom or 4 psi x 40 inches x 200 inches equals 16000 pounds. Adding 1000 pounds for the weight of the dome gives 17000 pounds total. This load will be resisted by a uniform tensile stress around the circumference of the side of the dome. The total cross-sectional area of the dome's side is (reference (1)):

$$Area = \frac{\pi}{2a}(3a^2 + b^2)t \tag{24}$$

where

a = semi-major axis = 50 inches;

b = semi-minor axis = 20 inches;

t = thickness of shell = 5/8 inch

Substituting into eq. 24:

$$Area = 156 \ in^2 \tag{25}$$

The tensile stress caused by the inertia loading is:

$$\sigma_{Tensile} = 17000 \ lbs / 156 \ in^2 = 109 \ psi$$

This stress has a factor of safety of 33. (26)

Buckling Resistance of Dome Sides to Impact Loads

The sides of the dome will resist only the buckling load during the slam condition. It is assumed that the lateral pressure loading is negligible. The basic theory for determination of the critical buckling stresses in plates with or without stiffeners is given in reference (o).

Unstiffened Simply Supported Curved Panel

The simplest analysis concerning the buckling resistance of the dome side is its consideration as a long simply supported curved plate, loaded along the larger length. From Fig. 1, the radius of the curved side is shown as 90 inches. From reference (o), the critical stress for a curved panel is given as:

$$\sigma_{Cr} = \frac{E H}{a \sqrt{3(1-\nu^2)}} \tag{27}$$

where: E = modulus of the plate = 2×10^5 psi

H = thickness of plate = 5/8 inch

a = radius of plate = 90 inches

ν = Poissons ratio = 0.45

Therefore:

$$\sigma_{Cr} = \frac{2 \times 10^5 \ psi \ (5/8 \ in)}{90 \sqrt{3(1-0.45^2)}} = 903 \ psi \quad * \tag{28}$$

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* For plates with supported edges (i.e., no edge free), the simply supported condition gives the lowest critical buckling stress. Use of the simply supported case gives a conservative estimate of the dome side strength.

The applied stress on the side panel will be calculated using the 7 psi criteria for slam loads. The total bottom area of the dome is approximately 40 inches by 100 inches, or 4000 in^2. The total impact load is:

$$F_{impact} = 7 \text{ psi} \times 4000 \text{ in}^2 = 28000 \text{ lbs} \tag{29}$$

The applied stress to the dome sides is the impact load divided by the cross-sectional area of the dome side:

$$\sigma_{applied} = \frac{28000 \text{ lbs}}{156 \text{ in}^2 \, *} = 180 \text{ psi} \tag{30}$$

The applied impact stress is far lower than the critical buckling stress. The factor of safety is 5.

<u>Area IV - Mounting Flange for Bolting</u>

The bolting technique is illustrated in Fig. 2. The technique utilized is similar to steel dome installation, and a fairing strip must also be applied with the polypropylene dome. Prior to investigating the strength of the flange, an analysis on the bolt strength will be made.

<u>Strength of Attachment Bolts</u>

Loading condition Type E, inertial loading will stress the bolts in tension. Equation (26) shows that the dome sides will experience a tensile stress of 109 psi through inertia loading. For a shell thickness of 5/8 inch, a load of 68 pounds per dome circumferential inch will be applied. The maximum stressed bolt near the transducer centerline will carry a load of approximately:

$$P_{bolt} = 68 \text{ lbs/in} \times 13 \text{ inches} = 884 \text{ lbs} \tag{31}$$

The bolt size is 7/8 inch except near the transducer, where it is 3/8 inch, see Fig. 2. The 3/8 inch bolt has a cross-sectional area of 0.068 square inches. Assuming a tensile working stress of 20000 psi, the bolt can carry:

$$P_{bolt} = 20000 \text{ psi } (0.068 \text{ in}^2) = 1360 \text{ lbs} \tag{32}$$

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* See equation (25).

With an applied load of 884 pounds, the factor of safety on bolt tension is: 1.5.

Shear Punchout of Plastic Support Beneath Bolt Head

Loading condition, Type E, inertial loading will stress the flange material through which the bolts penetrate. The bolt passes through a 1 1/4 inch thick section, t, see Fig. 2. The outer radius, r, of the bolt head is 3/4 inches. The shear punchout will probably occur through the 1 1/4 inch polypropylene at a section tangent to the outer bolt radius. This shearing area is:

$$\text{shear area} = 2\pi r t = 2(3.14)(3/4\text{ in})(1\,1/4\text{ in}) = 5.89\text{ in}^2 = A_s \qquad (33)$$

With an applied load of 884 pounds, the maximum shear punchout stress is:

$$\tau_{max} = P/A_s = 884\text{ lbs}/5.89\text{ in}^2 = 150\text{ psi} \qquad (34)$$

Assuming that the maximum allowable shear stress is 0.6 times the allowable tensile stress, i.e., 0.6 times 3600, or 2160 psi, the factor of safety against a shear punchout is 14.4.

Flexural Stress at Flange Support

From a type B loading, sideways form drag forces during a turning maneuver at 35 knots, flexural stresses will occur at the mounting flange of the dome. Assuming the side of the dome to act as a rectangular plate with three edges simply supported, and one edge built in, the maximum moment per unit width occurs at the built in edge and is equal to (reference (k)):

$$M_{max} = 0.122\, q\, b^2 \;;\; \text{for } b/a = 1/3 \;;\; \nu = 0.3 \qquad (35)$$

where

q = the uniform pressure load = 25 psi

b = the height of the dome side plate panel of 5/8 inch thick polypropylene = 33 inches

Substituting into equation (35):

$$M_{max} = 0.122\,(25\text{ psi/in})(33\text{ in})^2 = 3321\text{ lb-in/in} \qquad (36)$$

The maximum bending stress is given in reference (k) as:

$$\sigma_{max} = \frac{6 M_{max}}{h^2} \quad (37)$$

where h = thickness of the plate = 5/8 inch

Substituting into equation (37):

$$\sigma_{max} = \frac{6\ (3331)\ lb\text{-}in/in}{(5/8)^2} = 66,420\ psi \quad (38)$$

The abnormally high stress resulting at the flange juncture to the skirt is not surprising, since this area was also found in the steel domes to be critically stressed, see reference (i). To reduce the magnitude of bending stress, increased thickness of polypropylene dome shell would be necessary. This procedure would lead to degradation of the sonar capability. The dome as designed in Figs. 1 and 2, and analysed in this memorandum, does not have the structural strength to resist a 35 knot, 10 degree yaw condition of design. Area "A", Fig. 2, is also a critical understrength section of the flange.

The author concludes that

CONCLUSION

Under a 35 knot, 10 degree of yaw maneuver, very large bending stresses result at the dome flange connection to the skirt. These stresses are calculated to be on the order of 66,000 psi, far in excess of the strength of the polypropylene material. Under all other conditions of loading, the dome appears to be adequate for service. Creep or long time loadings have not been considered in the analyses. He recommends:

RECOMMENDATIONS

1) - The present design of the dome be rejected. and

2) - Additional design studies be performed. Consideration should be given to use of a composite or sandwich structure up at the flange portion, as a means of providing additional strength. The effect on the sonar should, of course, be evaluated.

MATTHEW BORG
Mechanical Engineer

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(b) Progress Report for Sonar Domes, Contract No. NObsr-42120, Report No. 2593-1, Edo Corp., College Point, New York, 28 May 1948 (CONFIDENTIAL) USL Accession No. 12874-1 (See also Accession Nos. 12562 and 13012)

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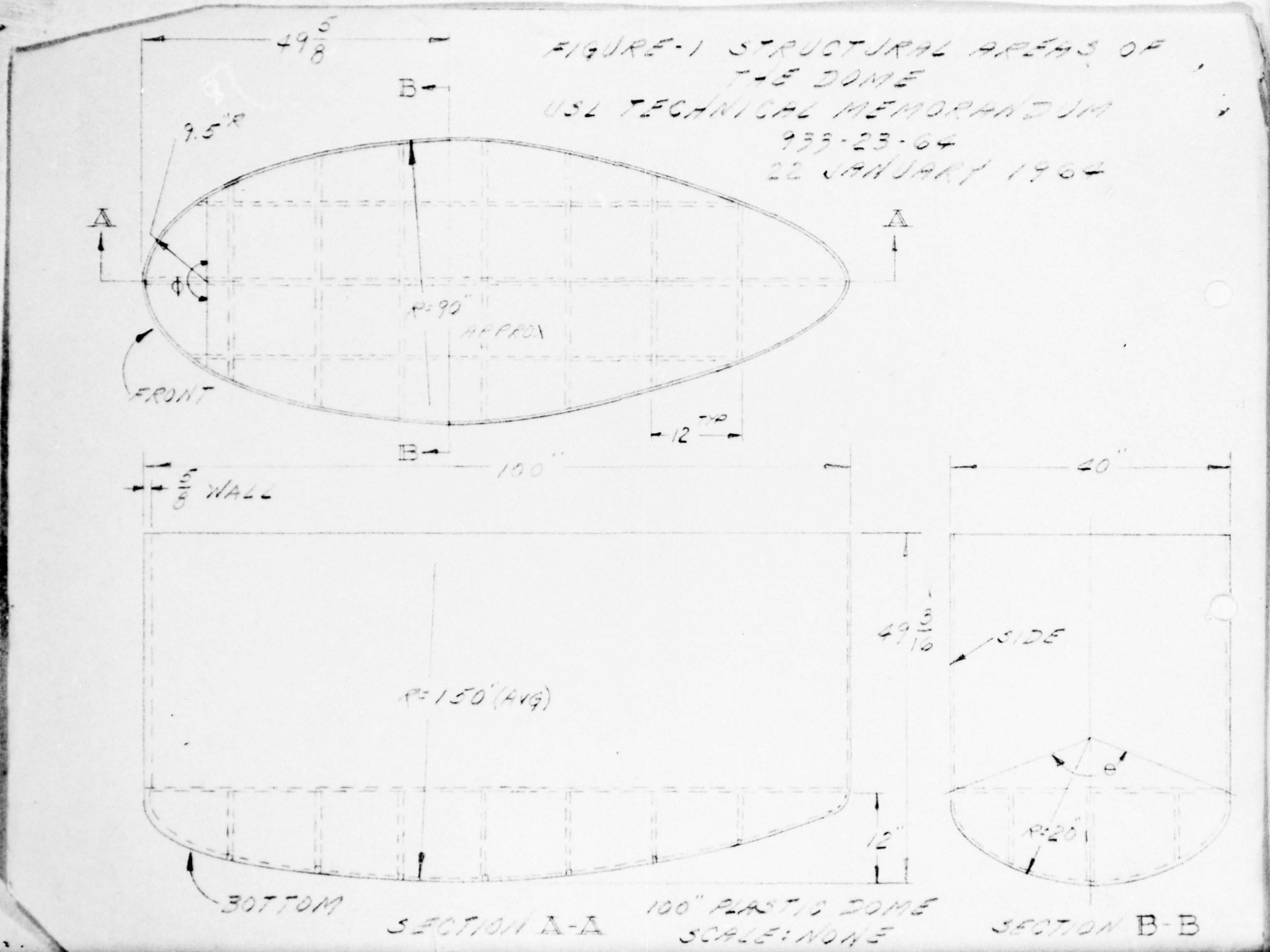

FIGURE-1 STRUCTURAL AREAS OF THE DOME
USL TECHNICAL MEMORANDUM
933-23-64
22 JANUARY 1964
49 5/8
B
9.5"R
A
A
R=90"
APPROX
FRONT
12 TYP
B
100"
5/8 WALL
40"
49 5/16
SIDE
R=150"(AVG)
θ
R=20"
12"
BOTTOM
SECTION A-A
100" PLASTIC DOME
SCALE: NONE
SECTION B-B

NOTE:
USE ARMSTRONG ARMALOY BOX SOCKET WRENCH #7039-A (OR EQUIV) NUT SIZE $1\frac{5}{16}$ + $1\frac{1}{2}$ ACROSS FLATS. 19" OVERALL

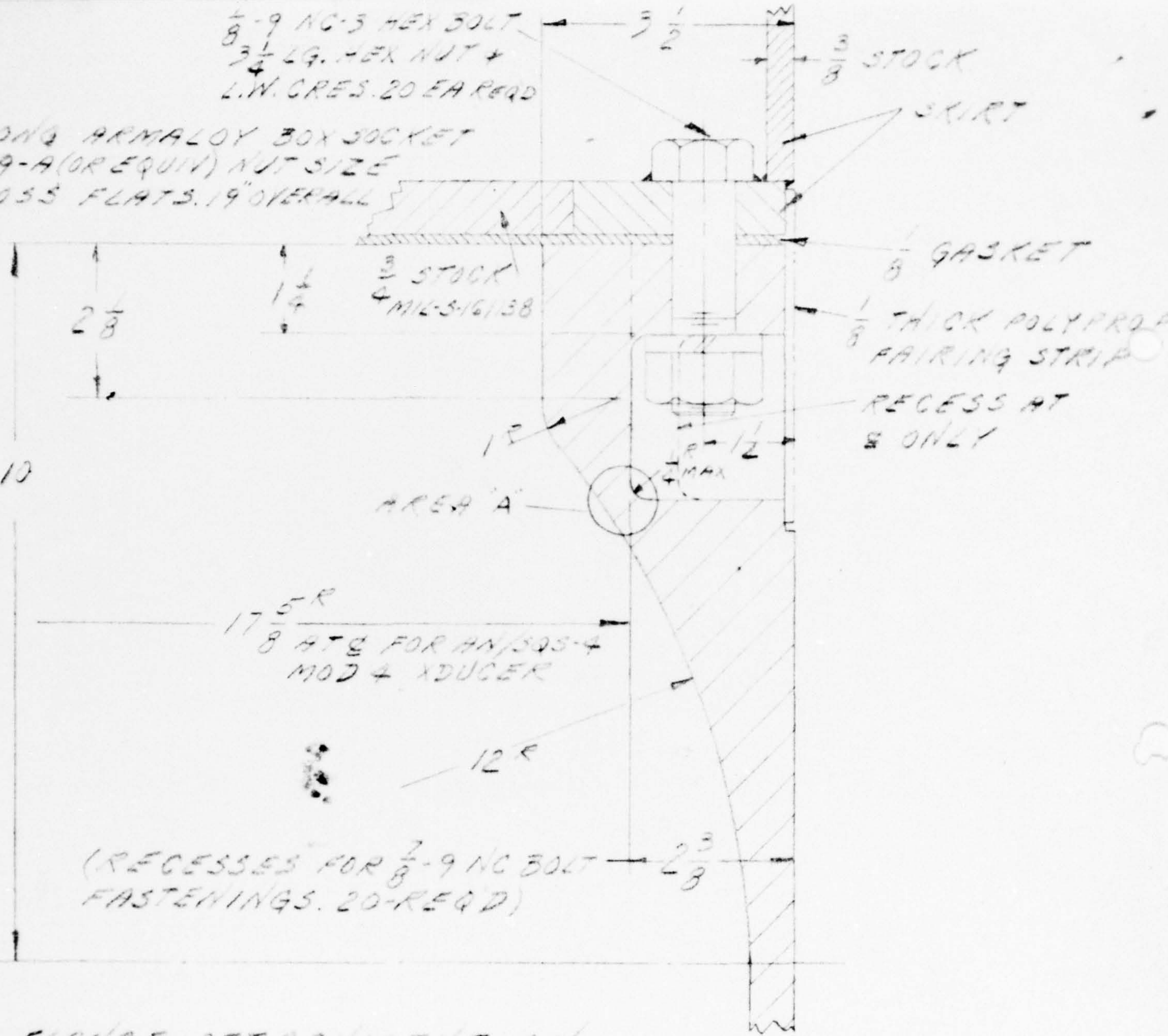

FIG-2 FLANGE ATTACHMENT ON TOP OF DOME

USL TECHNICAL MEMORANDUM
933-23-64 22 JANUARY 1964

Distribution List

Code 100
Code 101
Code 900
Code 900A
Code 900C
Code 930
Code 930S (3)
Code 932
H. Phelps
M. Borg
Code 933
Code 933.1
Guy Williams
Code 904
Code 902
Code 904.2 (5)

External

BUSHIPS (Code 689C) (3)